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ADVANCED POWER MANAGEMENT OF A TELEHANDLER USING ELECTRONIC LOAD SENSING

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Abstract: New possibilities within electronic control of mobile hydraulic systems are becoming available as hydraulic components are implemented with more electrical sensors and actuators.
This paper presents how the traditional hydro-mechanical load sensing (HLS) control of a specific mobile hydraulic application, a telehandler, can be replaced with electronic control, i.e. Electronic Load Sensing (ELS). The motivation for ELS is the potentials of better dynamic performance and system utilization, along with reduced mechanical complexity by transferring features as pump pressure control, flow-sharing, power-sharing, anti-stall and high pressure protection to electronic control. These features are integrated into the developed control structure, which is implemented and tested.
The ability of electronically controlling the position of the swash-plate in a variable-displacement pump is an essential part of the developed control solution. Hence, the development of a control structure for electronic control of a variable-displacement axial piston pump using a three-way servo valve is also treated.

Keywords: Mechatronics, Hydraulics, Fluid Power, Control, Electronic Load Sensing, Power Management

I INTRODUCTION

Today, the open circuit hydraulics on most mid and higher end mobile machinery utilizes a hydro-mechanical adaptable flow and pressure control scheme known as Hydraulic Load Sensing (HLS). Here the pump pressure is continuously controlled to be a preset margin higher than the highest load pressure.

The HLS-control scheme is implemented by controlling the displacement of a variable pump using a system of pilot-lines, pressure compensators and shuttle valves. Features like high-pressure protection, load-independent flow-control of actuators, flow-sharing, toque-limitation etc., are also implemented using these hydro-mechanical components.

The success of HLS systems is due to their robust components and improved energy efficiency compared to constant pressure and/or constant flow systems. The drawbacks of HLS is a high commissioning effort, as poorly designed HLS systems easily become oscillatory or even unstable. Due to the dynamical complexity of HLS systems, tuning is based upon experience and trial-and-error, and the tuning itself is cumbersomely performed by adjusting or changing springs, orifices, spool design, hose volumes, etc. The complexity and inflexibility of hydro-mechanical control also imposes restriction on making more adaptable and intelligent control solutions.

The introduction of electronically controlled components into the open circuit systems offers the desired increase in flexibility and adaptability in the control, as digital control can be utilized. Hence, improved system utilization and dynamic performance could potentially be obtained, along with faster commissioning as changing control parameters is effortless. The potential of electronic control in open circuit systems is also becoming recognized, and is supported by more stand-alone electronic controlled components being introduced into the market.

When moving to computer controlled components, a range of control concepts is available, from making an electronic analogy of the current HLS system to using a different control topology. The research conducted in electronic open circuit control mainly falls into three groups, Electronic Load Sensing (ELS), Sum-of-Flow control (SummenStromRegelung, SSR) and Electronic Flow Matching (EFM).

The ELS control topology is treated by Esders in [1] and Langen in [2], and is the electronic analogy to HLS. The LS pressure is electronically sensed and an electronic pump pressure controller operates the pump based on the LS-pressure and a pump pressure feedback. Regarding valve control, Esders found that by utilizing the pressure measurements the valve's main spools could perform electronic pressure compensation, thereby providing load independent proportional flow-control without the need of pressure compensators. This was also used in [3], where it was shown to be equivalent to a static feedback linearisation of the valve.

Taking another approach, the SSR control topology operates the pump without pressure feedback, but controls the pump to provide the sum of the requested flows. This can be performed in either a open or closed loop manor. The closed loop SSR control was investigated by Zähe in [4], where the velocities of the actuators constitute the main feedback signals for pump and valve control.

In [5] an open loop version of SSR is investigated, where pressure sensors are equipped on all valve ports. The pump displacement is controlled to give the sum of flow references, the control valve of the highest loaded consumer is completely opened and the flow to the lower loaded functions are controlled by their control valves using electronic pressure compensation. Hence, in this control scheme the pump is operated in pure feed-forward mode.

The open and closed loop SSR have the highest possible efficiency as no LS margin is present.

The EFM control scheme is closely related to SSR, but focus on an implementation with a minimum of electronics and sensors. The pump is operated in pure feed-forward mode, where the swash-plate is positioned to deliver the sum of flow demands. A traditional flow-sharing valve distributes the flow. The concept is covered in [6], [7] and [8]. The flow-sharing nature of the valve prevents and unstable pressure build up, as any flow oversupply is mutually distributed amongst the active consumers.

EFM is also investigated using a non-flow-sharing valve in [9] and [10]. However, secondary compensators and/or compensator position sensors are added to the system in order to provide feedback for stabilizing the system.

In the work presented in this paper, the goal is to obtain complete electronic control of open circuit systems in mobile hydraulics. The control philosophy is to:

- Transfer as much as possible of the hydro-mechanical control to electronic control, reducing the mechanical complexity to a minimum.
- Do system control with intelligent machine power management utilizing engine information.
- Have a general control structure applicable for mobile hydraulic systems, which is designed and tuned using system information.

The control strategy moves away from the EFM principle of a minimum of electronic sensors, but moves in the direction of the ELS and open loop SSR, but with added overall system control and power management. The closed-loop SSR is not included, as the project is avoiding the need of position or velocity sensors on the application's actuators.

The work presented in this paper is a result of the first feasibility testing. Here a developed distributed control structure is tested on a mobile machine with electronic controlled open circuit components. The control is in its feasibility stage, hence only an ELS based control topology is investigated, as it is considered a basic and required control mode.

II SYSTEM DESCRIPTION

The application used for ELS feasibility testing is a Merlo P 35.11 EVS telehandler from 1997. As seen in Fig. 1, the machine has been re-equipped with electronic controlled open circuit components and a diesel engine with CAN interface. The engine transmits engine speed ω_e and current torque load τ_{actual} every 20 ms.

All the control is handled by a central computer from Speedgoat with CAN and analog I/O interface. The software platform used on the Speedgoat for implementing control is the xPC Target™ environment for MATLAB/SIMULINK by The MathWorks.

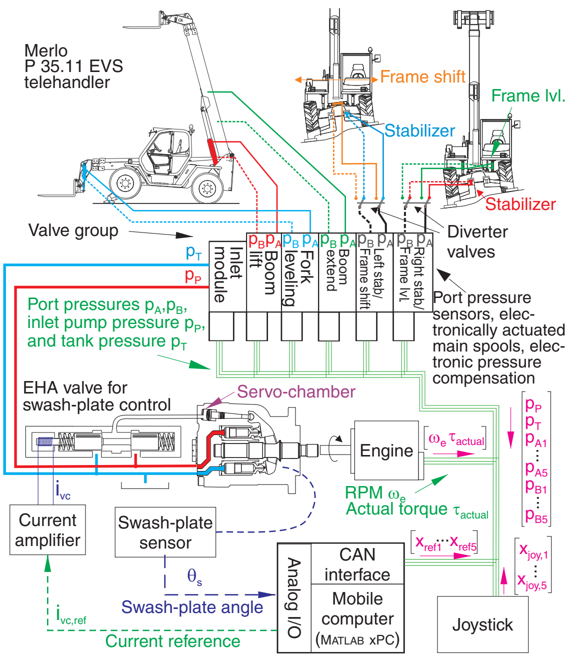

Figure 1: Telehandler with ELS

The valve-group is equipped with pressure sensors on all ports, and uses electronic pressure compensation for load-independent flow control. A fluid temperature sensor, a pump pressure sensor and a tank pressure sensor are mounted at the valve-inlet. The valve-group receives reference spool position on the CAN bus from the Speedgoat.

Flow reference signals from the operator are received from a CAN joystick transmitting every 20 ms. The pump is a modified Sauer-Danfoss Series 45 J-frame 60cc variable-displacement axial-piston pump. The pump has been equipped with a swash-plate position sensor, and the traditional hydro-mechanical LS-control housing has been replaced with an electronically actuated three-way valve (EHA-valve). The EHA valve is utilized for providing electronic swash-plate control, where the developed control algorithm is described in the following.

III Swash-Plate Control

The voice-coil actuated EHA-valve used for controlling the pump is based on the EHA valve used in [11] for testing electronic control of a Sauer-Danfoss Series 45 H-frame pump. The voice-coil actuator gives a force proportional to the applied current. A cascade control was utilized in [11], consisting of a current loop PI-controller, a spool positioning loop with a PID-controller for the EHA valve, and a PID-controller for the swash-plate positioning loop.

In this paper, a swash-plate control algorithm not requiring the measurement of the EHA spool position as in [11] is developed. In [11] the EHA positioning loop serves for increasing the bandwidth of the spool and to suppress the disturbance from flow forces, which tends to close the valve. To replace the position feedback, the strategy is to implement a spool position estimator.

In the system setup in Fig. 1 a current amplifier with internal current controller is utilized. Note that the EHA valve is equipped with a spool position sensor for identification and verification purposes.

If flow-forces are omitted and the friction modeled as of viscous type, which is valid as a dither signal is applied to the spool, the following linear second-order model is obtained for the spool,

$$\frac{x_{\mathrm{EHA}}}{i_{\mathrm{ref}}} = \frac{\frac{k_{\mathrm{f}}}{m_{\mathrm{s}}}}{s^2 + \frac{\mu}{m_{\mathrm{s}}}s + \frac{k_{\mathrm{s}}}{m_{\mathrm{s}}}} \tag{1}$$

where k_{f} is the voice-coil force-factor, m_{s} is the spool mass, μ is the viscous friction coefficient and k_{s} is the equivalent spring stiffness of the two springs.

To identify the friction coefficient, a ramped square-wave current is applied to the voice-coil with the spool in oil, but at zero pump pressure. The response of the valve is seen in Fig. 2, where μ in (1) has been identified, yielding the following second-order characteristics: $\omega_{\mathrm{n}} = 317\,\frac{\mathrm{rad}}{\mathrm{s}}$, $\zeta = 6$ and a DC-gain of $K = 4.6\,\frac{\mathrm{mm}}{\mathrm{A}}$.

Hence, the spool is extremely over-dampened, with a rise-time of approximately 80 ms. In comparison a traditional LS-pump has an outer swash-plate positioning loop with a rise-time in the range of 50 ms to 150 ms, making the EHA valve too slow.

To increase the bandwidth, (1) is synthesized to provide a spool feedback signal. Equation (1) is however an open-loop estimator, unable to compensate for flow-forces. From measurements it has been seen that the flow-forces are able to pull the spool up to 1.5 mm away from the position estimated by (1), degrading the value of the estimator, especially when comparing to a maximum spool movement of -2 mm to 2.5 mm.

To solve this, the swash-plate position measurement is used for estimating the flow-force. By differentiating the swash-plate position, the swash-plate

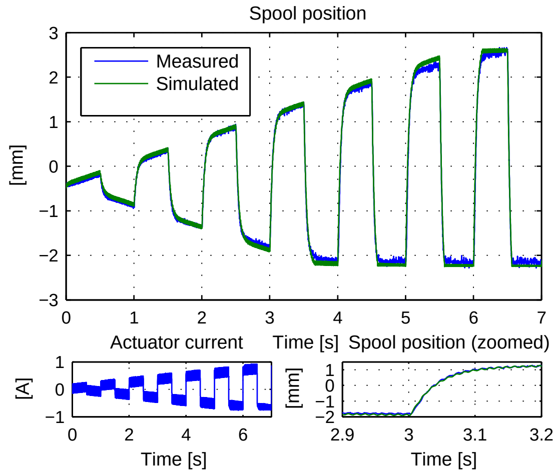

Figure 2: Valve dynamics and spool model.

velocity ω_{s} is obtained, which is approximately proportional to a flow through the EHA-valve $Q_{\mathrm{EHA}} = \omega_{\mathrm{s}} A_{\mathrm{DP}} L_{\mathrm{DP}}$, as the pressure build-up in the small servo-chamber volume can be neglected. The area A_{DP} denotes the servo-piston area and L_{DP} is the piston's moment-arm on the swash-plate.

By measuring how much the flow-forces makes the spool position divert from (1) for different flows, a map from Q_{EHA} to F_{Ff} has been made and implemented in the controller, see Fig. 3. With knowledge about the flow-force, the controller can map F_{Ff} into an equivalent current and use it as a feed-forward to decouple the effect of the flow-force. Resultantly, only the dynamics of (1) remains, making it a valid estimator, except at spool endstops.

Endstops are often modeled as the appearance of a large stop force proportional to the distance the object has passed its endstop x_{min} or x_{max}:

$$F_{\mathrm{endstop}} = \begin{cases} k_{\mathrm{e}}(x_{\mathrm{min}} - x_{\mathrm{EHA}}) & \text{if } x_{\mathrm{EHA}} < x_{\mathrm{min}} \\ k_{\mathrm{e}}(x_{\mathrm{max}} - x_{\mathrm{EHA}}) & \text{if } x_{\mathrm{EHA}} > x_{\mathrm{max}} \end{cases} \tag{2}$$

The gain k_{e} of the endstop can however easy result in the model showing limit cycles at the endstop.

To design k_{e} properly, the endstop function in (2) can be viewed as a feedback-controller, giving an input F_{endstop} to control the spool to x_{min} or x_{max} when these are exceeded. Using (2) as a controller for (1), the resulting closed loop system in (3) is found, where the gain k_{e} designed to give a proper endstop, which is a critical dampened system ($\zeta = 1$), as it settles the fastest with no overshoot.

$$x_{\mathrm{EHA}} = \frac{\frac{\frac{1}{m_{\mathrm{s}}}}{s^2+\frac{\mu}{m_{\mathrm{s}}}s+\frac{k_{\mathrm{s}}}{m_{\mathrm{s}}}}}{1+\frac{\frac{1}{m_{\mathrm{s}}}}{s^2+\frac{\mu}{m_{\mathrm{s}}}s+\frac{k_{\mathrm{s}}}{m_{\mathrm{s}}}}k_{\mathrm{e}}} = \frac{\frac{1}{m_{\mathrm{s}}}}{s^2+\frac{\mu}{m_{\mathrm{s}}}s+\left(\frac{k_{\mathrm{s}}}{m_{\mathrm{s}}}+\frac{1}{m_{\mathrm{s}}}k_{\mathrm{e}}\right)}$$

$$\Rightarrow 2\zeta\omega_{\mathrm{n}} = \frac{\mu}{m_{\mathrm{s}}} \overset{\zeta=1}{\Rightarrow} k_{\mathrm{e}} = \frac{\mu^2}{4m_{\mathrm{s}}} - k_{\mathrm{s}} \tag{3}$$

The endstop is implemented and used in the simulation showed in Fig. 2. This endstop also has the advantage that if defined by model parameters, it always fits the model despite parameter changes.

A EHA controller is implemented using the estimator, along with a feed-forward part to give the steady-state current for a given spool position. As the spool is very over-dampened a simple P-controller can be used to increase the bandwidth without making the system unstable. Using (1), a proportional gain $k_{\text{p,EHA}} = 1.5\ \frac{\text{A}}{\text{mm}}$ has been determined, yielding a bandwidth of 100 Hz and a rise-time of 10 ms, while providing a damping factor of $\zeta \approx 2$. With this setup, the EHA-controller can handle steps up to approximately 0.5 mm without actuator saturation.

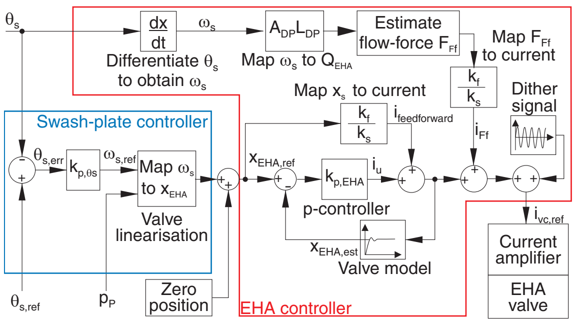

Figure 3: Swash-plate controller

The developed EHA-controller is used as an inner loop for the swash-plate controller. As the EHA-valve is critical lapped, a natural integrator is always present from spool reference to swash-plate position. Consequently, no integrator-term is required in the swash-plate controller.

The gain from spool reference $x_{\text{EHA,ref}}$ to swash-plate velocity ω_{s} is non-linear dependent on the pump pressure p_{P} and the pressure in the servo-chamber p_{DP} according to the orifice equation. Furthermore, a greater pressure loss is present across the spool when de-stroking due to the pump geometry, which approximately gives the static relation $p_{\text{DP}}{=}\frac{1}{3}p_{\text{P}}$. As the area characteristic of the spool is known and the pump pressure is measured, these can be used to statically cancel out the non-linearities by implementing an inverse orifice equation as in electronic pressure compensation by [1] and [3]. Hence, a mapping is performed from a flow Q_{EHA} to a spool position.

As the flow Q_{EHA} corresponds to a swash-plate velocity $\omega_{\text{s}}{=}\frac{Q_{\text{EHA}}}{A_{\text{DP}}L_{\text{DP}}}$, the control input to the system actually becomes a swash-plate velocity reference $\omega_{\text{s,ref}}$. Thus, the last loop of the swash-plate controller reduces to a P-controller, sending out a swash-plate velocity reference $\omega_{\text{s,ref}}$. This also emphasizes that no integrator is required in this control loop. The complete swash-plate controller is seen in Fig. 3.

IV System Control

The over-all control topology chosen for the open circuit system is ELS, as pump pressure control is considered a basic and required functionality. Open-loop SSR for example has its limitation if consumers with unknown flow consumption are present, or highest loaded consumers have similar pressure levels.

The main ELS control consists of identifying the LS-pressure, add it with a margin and use it as a reference signal for a PID pump pressure controller as seen in Fig. 4. Note that pump pressure controller is closed around the pump pressure measured at the valve inlet. This gives the advantage that the controller compensates for the pressure loss from pump to valve-group, making a less conservative choice of LS-margin possible compared to HLS.

The gain of the pump increases with engine speed, as the pump requires a smaller swash-plate angle to deliver the same amount of flow. To statically cancel out this non-linearity, the pump pressure controller is designed to always send out a flow reference, which is then mapped into a swash-plate angle using the engine speed, along with compensating for the volumetric efficiency of the pump.

The flow consumed by the valve group can be viewed as a disturbance to the pressure controller, which has to continuously adapt to the flow consumption based on a pressure error. The flow disturbance is however known in advance from the joystick inputs, which gives the flow references. These can be summarized to a feed-forward signal Q_{FF} to the pump, thereby decoupling the disturbance. Resultantly, the pressure control loop only has to make small flow corrections $\tilde{Q}_{\text{ref}}$ to maintain the correct pump pressure. A similar approach was used in [12].

An extra advantage of electronic pump pressure control is that high pressure protection in the system is simply implemented as a saturation limit on the pump pressure reference, and the standby-pressure is determined by a the lower limit saturation. Note that in ELS the standby pressure is separated from the LS-margin setting, where these are normally equal in HLS due to the hydro-mechanical implementation.

The above constitutes the basics of the controller. The following describes how flow saturation, power saturation and actuator endstops are handled.

A. *Flow and Power Saturation*

Flow saturation occurs when the operator request more flow than the pump can deliver. In HLS system with non-flow-sharing valves, the highest loaded consumer would be starved and the LS-margin lost. In HLS with flow-sharing valves, the available flow would be mutually distributed, but with reduced LS-margin, and hence reduced control precision.

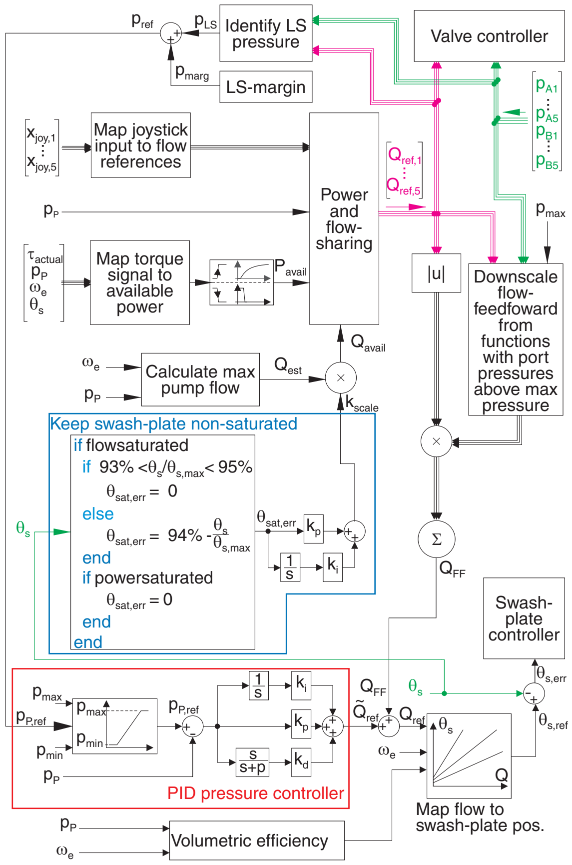

Figure 4: ELS control structure

In the ELS system the engine speed is used for estimating the available flow. Hence, if the operator's flow request exceeds the available flow, a mutual down scaling is performed on the references by a flow-sharing block, such that the sum is equal to the available flow. These scaled references are then send to the valve-group.

The sum of inaccuracies in estimated available flow and the valve-group flow control can however still result in trying to use more flow than available, thereby making the system fail to maintain the LS-margin.

To deal with this, two control strategies can be used for the pump pressure control during flow saturation. The first strategy is to disable the normal pressure controller, force the swash-plate to maximum angle, and then control the pump pressure by adjusting the available flow input Q_{avail} to the flow-sharing block. This however requires a shift between controllers, and the gain from Q_{avail} to p_{P} is very dependent on the changing valve dynamics. It might also give small ripples in the valve flows.

The second strategy is to realize that the system has two control inputs, the swash-plate angle and the flow-consumption of the valves Q_{avail}, to control one output, the pump pressure. For such control problems mid-range control, see e.g. [13], is often effective, which consists of two controllers: A main controller controlling the output with one of the inputs, and a secondary controller keeping the main controller out of saturation using the second input, i.e. keeping the main controller in mid-range.

Applied to this problem, the swash-plate angle is used to control the pump pressure, such that the same pressure controller is always active. The secondary controller is then used to scale the valve-group's flow-consumption Q_{est}, such that the swash-plate never saturates. As it is important to utilize the pump capacity, the secondary controller is set to keep the swash-plate angle between 93% to 95% of maximum during flow saturation, instead of having it strictly in midrange. The implemented "skewed" midrange strategy is seen in Fig. 4. As the secondary controller is implemted as a PI, the sytems also obtains the ability to continuously adapt to changing the flow charteristics of valves and pump due to e.g. fluid temperature.

Power-saturation might also occur, giving a risk of engine stall. To prevent this, the engine continuously transmits its current torque load to the ELS controller, where it is used for calculating the available power P_{avail}. Using the pump pressure, the maximum allowed pump flow without exceeding P_{avail} can be calculated as $Q_{\max} = P_{\text{avail}}/p_{\text{P}}$. Using the same flow-sharing algorithm as during flow saturation, flow references are scaled such that their sum never exceeds $Q_{\max}$. Note that in case of simultaneous power and flow saturation, the midrange-controller does not attempt to force the swash-plate to 95% of maximum. Finally, a non-linear filter is added to the P_{avail} to avoid limit cycles with the engine by ramping P_{avail} slowly up.

B. *Intelligent High Pressure Control*

When one of the telehandler's functions, for example boom lift, reaches its endstop, the LS-pressure rises until the pump-pressure limit is reached by the pump-pressure controller. In this case, the flow-feed forward Q_{FF} from the joystick to the pump ceases to be valid, as the flow reference to the function in endstop is not consumed. Thus, the pump-pressure controller now has to remove this excessive flow by integration. To improve the performance, the functions in endstop or otherwise blocked, can be identified by their port pressure reaching $p_{\max}$. Consequently, the flow feed-forward from these functions is scaled down when it approaches $p_{\max}$, making the flow feed-forward to the pump pressure controller valid again, and improves its performance. This is also implemented in Fig. 4.

This control feature completes the control design, which is implemented on the telehandler.

V RESULTS

In Fig. 5 a test of the swash-plate control algorithm is performed, where the load is a relief valve with a 50 bar setting. The swash-plate is initially tracking a 10 Hz sine-wave, and afterwards stepped from minimum to 95% of maximum stroke and back again. The destroke happens in 80 ms and the on-stroke is in 150 ms. Note that the on-stroke is from zero pressure.

The swash-plate tracks the sine-wave with a phase lack of 45°, but with full amplitude as seen in the zoomed view in Fig. 6. The plot also shows the control signals and the spool position. Notice that the estimated spool position shows the same dynamics as the estimated, and that the swash-plate velocity reference given out of the controller resembles the actual velocity, as the controller was designed for.

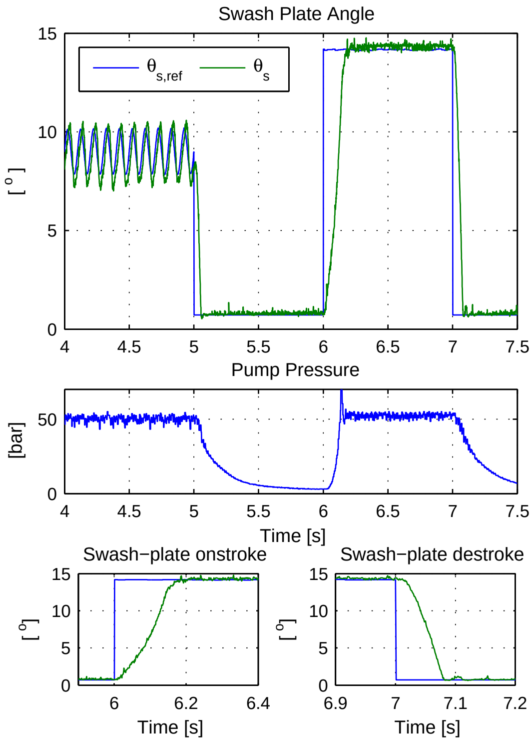

Figure 5: Swash-plate control test

In Fig. 7 a stabilizer is operated with the engine idling. The stabilizer is lowered to lift the machine, and then retracted to its initial position. The controller is configured to maintain an LS-margin of 20 bar which is maintained, and the swash-plate tracks the reference signal. At 8 s and 16 s the stabilizers reaches mechanical endstop, forcing the pump pressure to maximum. It is seen that the flow feed-forward Q_{FF} is reduced due to the intelligent high pressure protection, and that the pump pressure is maintained at its maximum pressure setting of 210 bar. At 11 s the machine is almost stalling, which can be seen

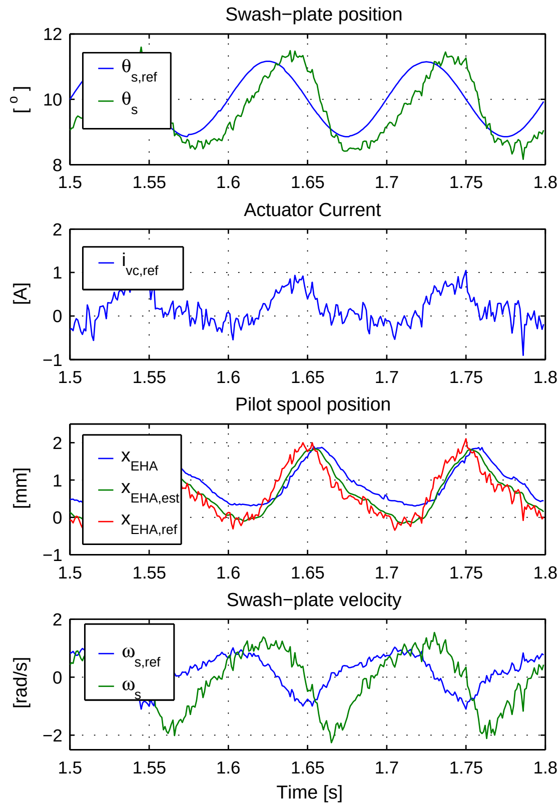

Figure 6: Swash-plate control details

by the engine speed dropping to 700 RPM and the available power P_{avail} going to zero. As a result, the flow feed-forward is eliminated by the system and the swash-plate de-stroked. Afterwards the system ramps up again as the engine recovers and P_{avail} rises.

In Fig. 8 the response time of the system is shown. The system is in standby, when a joystick flow reference is given. It is seen that the joystick flow-reference is directly feed-forwarded to a swash-plate reference, and the swash-plate has settled in 90 ms.

In Fig. 9 multiple consumers are operated simultaneously (boom, extension and fork), where the requested flow exceeds the available. As seen, the system scales down the flow references (the black curves in the three topmost plots), such that the flow feed-forward Q_{FF} is almost constant, except when the engine speed is increased at 8.5 s. As noticed the swash-plate is kept at 95% of max, which is approximately 16°. This enables the pump pressure controller to maintain the LS-margin of 20 bar at all time. The flow adaption graph shows the output of the mid-range controller for scaling the valve flow, such that the swash-plate is kept non-saturated.

In Fig. 10 the boom is operated, where the LS-margin has been lowered to 7 bar, while still having a standby pressure of 20 bar. As seen, the system is able to maintain the LS-margin and standby pressure, while providing a well-damped swash-plate control.

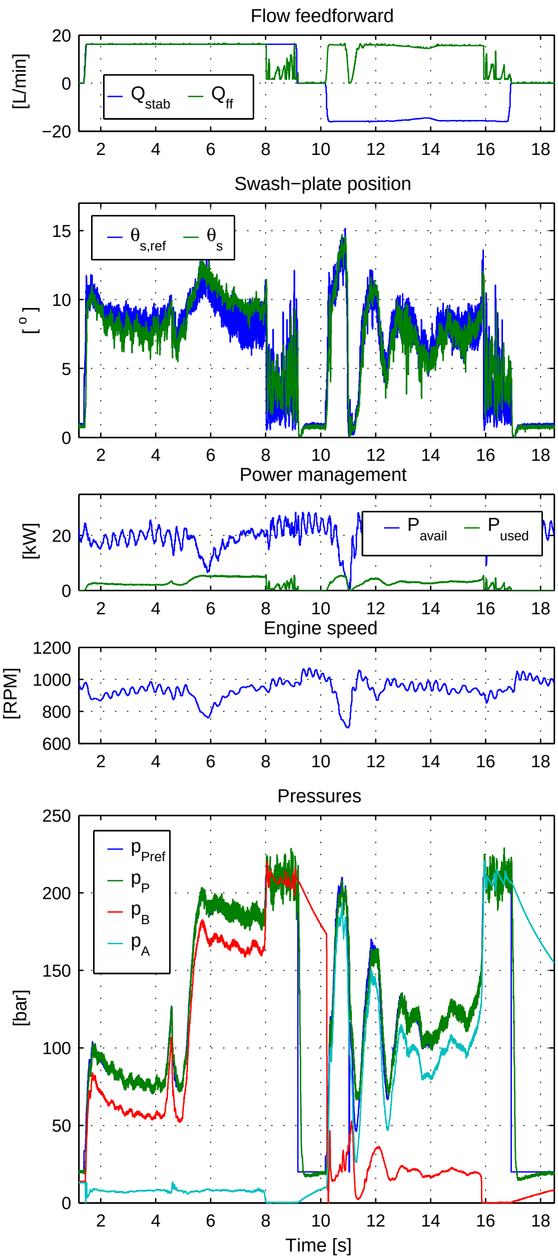

Figure 7: Stabilizer trajectory with engine idling

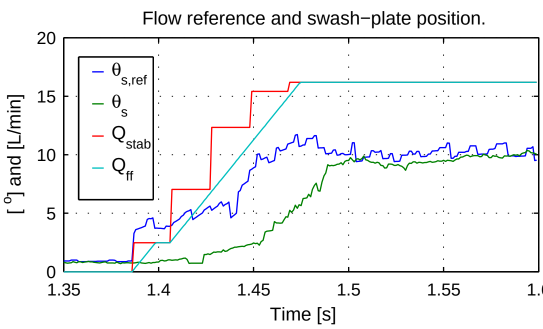

Figure 8: System response time, a zoomed view of Fig. 7 at time 1.35 s

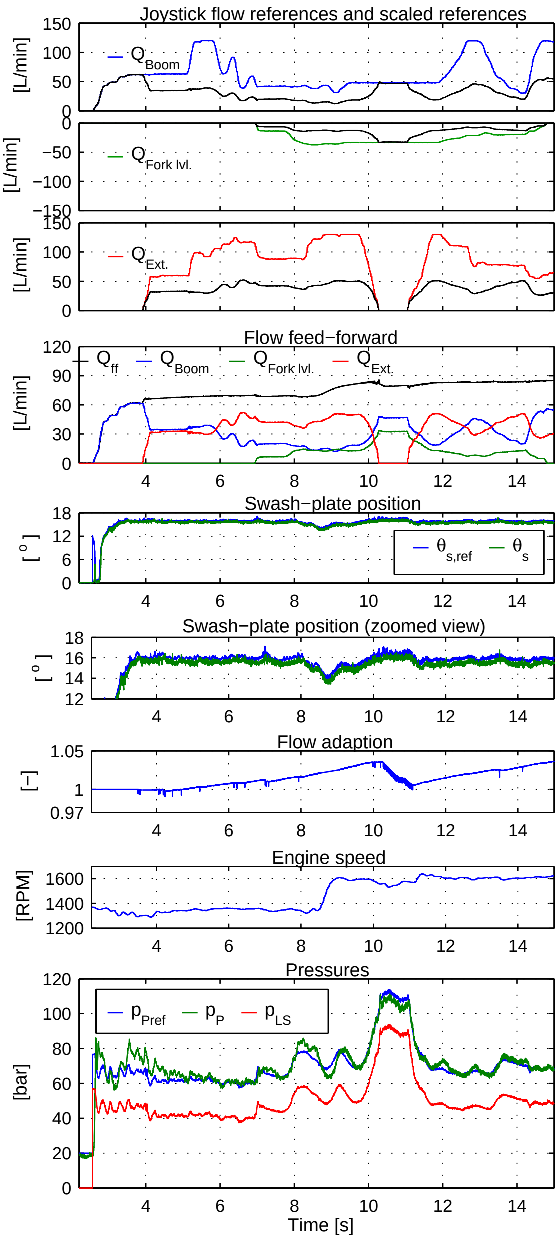

Figure 9: Test of flow-sharing

VI Discussion

The developed swash-plate controller for actuation of a pump using a three-way valve shows a bandwidth of at least 10 Hz, validating the control concept. This shows that it is possible to replace the spool position sensor of the EHA valve with an estimator, when the flow-forces are decoupled based on the swash-plate measurement for estimating flow through the valve. Based on the system performance it is concluded that the obtained bandwidth, and on-stroke and de-stroke times of the electronic controlled pump are sufficient.

The developed open circuit controller shows that it is possible to move features as pressure control,

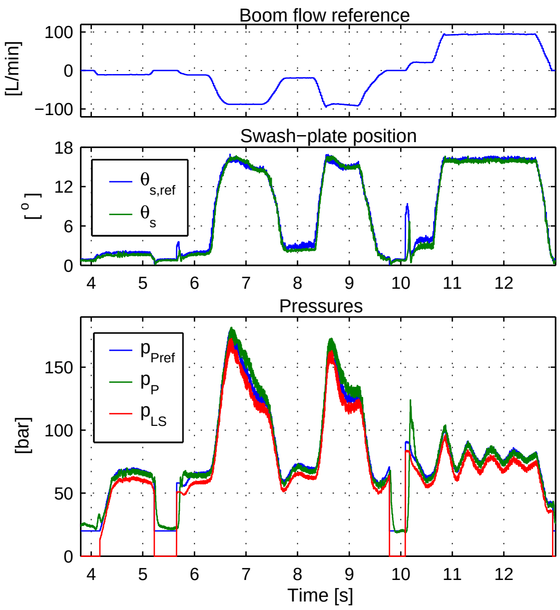

Figure 10: Boom trajectory with 7 bar LS-margin

flow-sharing, anti-stall, power sharing and maximum pressure limitation from hydro-mechanical control to electronic control, thereby also limiting the mechanical complexity of the system.

The developed control structure proves the effectiveness of an ELS control topology with a flow feed-forward from the joystick to the pump. This reduces the pump pressure controller to making small adjustment to the swash-plate in-order to maintain the desired pump pressure. The idea of eliminating the flow feed-forward from consumers reaching pressure limitation has also been shown to be a valid solution for maintaining the integrity of the flow feed-forward.

To elegantly handle flow-saturation and increase the robustness, the skewed-midrange topology used during flow saturation has been validated. It ensures that the swash-plate never saturates, and that 93%-95% of maximum flow capacity is alway utilized during flow-sharing. As a result, the same pump pressure controller is aways operational, thereby ensuring that the LS-margin is always maintained, and makes the system continuously adapt to the flow characteristic of the pump and valves.

Regarding the implemented control, one of the overall results of the paper is the usefulness of canceling out non-linearities using system information to create appropriate coordinate shifts, for example from flow references to spool positions.

Finally, the results show that using ELS the system can operated with an LS-margin down to 7 bar, while having a standby-pressure setting of 20 bar.

The generality of the developed control structure makes it reasonable to expect that it is applicable for other mobile hydraulic application than a telehandler.

REFERENCES

[1] H. Esders. Elektrohydraulisches Load Sensing für Mobile Anwendungen. PhD thesis, Teachnischen Universität Braunschweig, Germany, 1995.

[2] A. Langen. Experimentelle und Analytischen Untersuchungen an Vergesteuerten Hydraulisch-Mechanischen und Elektro-Hydraulischen Pumpenregulungen. PhD thesis, Rheinisch-Westälishen Technischen Hochschule Aachen, Germany, 1986.

[3] B.K. Nielsen. Controller Development for a Separate Meter-In Separate Meter-Out Fluid Power Valve for Mobile Applications. PhD thesis, Aalborg University, Denmark, 2005.

[4] B. Zähe. Energiesparende Schaltungen Hydraulischer Antrieb mit Veränderlichem Versorgungsdruck und ihre Regelung. PhD thesis, Rheinisch-Westfälischen Technischen Hockschule Aachen, Germany, 1993.

[5] H. Jongebloed, D. Büren, U. Völkel and C. Jabs. Energy-Saving Valve System for Mobile Applications - Load-Control-System (LCS). 4th International Fluid Power Conference, Dresden, Vol.2, 2004.

[6] C. Latour. Electrohydraulic Flow Matching: The next generation of load-sensing controls. Off Highway March, 2007.

[7] R. Finzel and S. Helduser. Energy-Efficient Electro-Hydraulic Control Systems for Mobile Machinery/Flow Matching. 6th International Fluid Power Conference, Dresden, 2008.

[8] M. Djurovic and S. Helduser. Elektrohydraulisches Load-Sensing. Ölhydraulik und Pneumatik, 11, 2004.

[9] M. Djurovic and S. Helduser. New control strategies for electrohydraulic load-sensing. Power Transmission and Motion Control (PTMC), 2004.

[10] K. Mettälä, M. Djurovic, G. Keuper and P. Stachnik. Intelligent Oil Flow Management with EFM: The Potentials of Electrohydraulic Flow Matching in Tractor Hydraulics. The Tenth Scandinavian International Conference on Fluid Power, SICFP'07, Tampere, Finland, 2007.

[11] M.R. Hansen, T.O. Andersen, H.C. Pedersen and F. Conrad. Feasibility Study of Electronic Load Sensing Concept for Hydraulic Variable Displacement Pump. , REM2006, KTH, Stockholm, Sweden, 2006.

[12] H.C. Pedersen. Automated Hydraulic System Design and Power Management in Mobile Applications. PhD thesis, Aalborg University, Denmark, 2007

[13] K.J. Åström and T. Hägglung. Advanced PID Control. ISBN/ID no. 978-1-55617-942-6, ISA, 2006.